REPORT No. 205

THE EFFECT OF CHANGES IN COMPRESSION RATIO UPON ENGINE PERFORMANCE

By STANWOOD W. SPARROW
Bureau of Standards

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SUMMARY

This report is based upon engine tests made at the Bureau of Standards during 1920, 1921, 1922, and 1923 and has been prepared for publication by the National Advisory Committee for Aeronautics. The majority of these tests were of aviation engines and were made in the altitude laboratory. For a small portion of the work a single cylinder experimental engine was used. This, however, was operated only at sea level pressures.

The report shows that an increase in brake horsepower and a decrease in the pounds of fuel used per brake horsepower hour usually results from an increase in compression ratio. This holds true at least up to the highest ratio investigated, 14 to 1, provided there is no serious preignition or detonation at any ratio. To avoid preignition and detonation when employing high compression ratios it is often necessary to use some fuel other than gasoline. It has been found that the consumption of some of these fuels in pounds per brake horsepower hour is so much greater than the consumption of gasoline that it offsets the decrease derived from the use of the high compression ratio. The changes in indicated thermal efficiency with changes in compression ratio are in close agreement with what would be anticipated from a consideration of the air cycle efficiencies at the various ratios. In so far as these tests are concerned there is no evidence that a change in compression ratio produces an appreciable, consistent change in friction horsepower, volumetric efficiency, or in the range of fuel-air ratios over which the engine can operate. The ratio between the heat loss to the jacket water and the heat converted into brake horsepower or indicated horsepower decreases with increase in compression ratio.

INTRODUCTION

Interest in this subject dates from 1882 when Sir Dugald Clerk showed the efficiency of the Otto cycle to depend solely upon the expansion ratio. Compression and expansion ratios are equal in conventional internal combustion engines and it is customary to speak merely of a change in compression ratio when both ratios are changed. When one ratio is changed and not the other special attention is drawn to that fact. This practice is followed throughout this report. Presumably early attempts to use high compression ratios failed because of what are now known to be preignition and detonation. Inasmuch as there was then no great demand for engines of extremely high thermal efficiency there was little incentive toward the overcoming of these difficulties. With the advent of the aviation engine conditions were reversed. Every effort was made to increase the thermal efficiency and the ratio of engine power to engine weight and to that end means were sought which would permit further increase in compression ratio.

The first attention given to this subject at the Bureau of Standards consisted in comparative tests of the performance of an aviation engine when equipped with pistons having compression ratios of 4.7, 5.3, and 6.2.[1] These tests clearly indicated the desirability of employing high compression ratios but were not sufficiently complete to serve as a basis for predicting the

[1] Compression ratio is the ratio of the sum of the piston displacement and clearance volume to the clearance volume.

probable gain from a given increase in compression ratio or to indicate the extent to which the compression ratio might be increased with profit.

To obtain such information a program was formulated which called for tests of an aviation engine when equipped with pistons giving compression ratios of 5.3, 6.3, 7.3, and 8.3, respectively. These tests were to be conducted over a range of air pressures and temperatures covering atmospheric conditions between sea level and an altitude of 25,000 feet. Performance was to be measured over a wide range of fuel air ratios, engine speeds, and engine loads in order that the results obtained might be of the widest possible application.

When the tests specified in the above program had been completed it was decided to extend the work to higher compression ratios by means of single cylinder engine tests. Two British investigators, Gibson and Ricardo, already had published considerable information derived from tests of single cylinder engines with various compression ratios and a sufficient number of tests had been made upon the single cylinder engine at the Bureau of Standards to show the extent to which its performance agreed with that of multi-cylinder engines. While the single cylinder engine tests were in progress two types of aviation engines were being tested in the altitude laboratory. At the request of the Bureau of Aeronautics of the Navy each of these types was tested with two compression ratios. The information derived from these and the previously mentioned tests forms the basis of the following report.

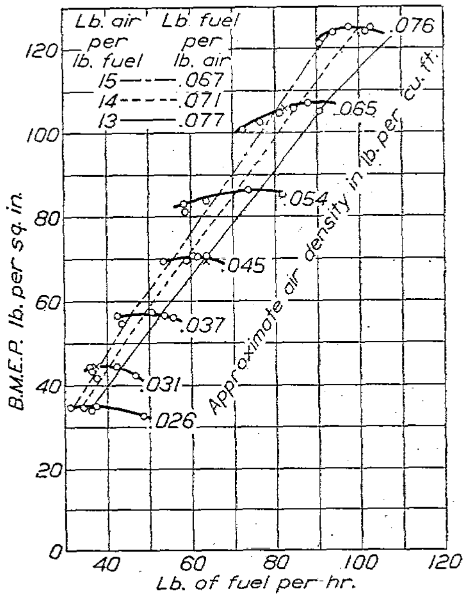


FIG. 1.—Test 163. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders; compression ratio, 5.3 : 1; air temperature, +10° C.; full throttle, 1,600 R. P. M.

ENGINES USED IN TEST

The performance of *aviation* engines as affected by compression ratio was of chief interest in this investigation. Consequently tests were for the most part made in the altitude laboratory where atmospheric conditions similar to those prevailing at high altitudes could be produced. For the major series of tests an airplane engine of a type which had given excellent performance in service was employed. This engine was of the 8-cylinder, water-cooled, V type, and had a bore of 4.72 inches and a stroke of 5.12 inches. Four sets of pistons were provided, giving compression ratios of 5.30, 6.30, 7.30, and 8.30. The differences in compression ratio were the result of differences in the amount of crowning given to the piston head.

In connection with another investigation, altitude chamber tests were made of an 8-cylinder engine of 5.51-inch bore and 5.91-inch stroke and of a 6-cylinder engine of 6.62-inch bore and 7.50-inch stroke. The performance of the former engine was measured with compression ratios of 6.50 and 5.40, whereas the latter was tested with ratios of 6.50 and 5.50. For extending the investigation to higher ratios a single cylinder experimental engine of 5-inch bore and 7-inch stroke was utilized. This was operated (at sea level only) with compression ratios of 5.4, 6.1, 7.2, 9.2, 11.5, 14.0.

GENERAL METHOD OF OBTAINING AND PLOTTING DATA

The general method of test was to measure the performance of the engine when operating with the various compression ratios over a wide range of typical service conditions. Figure 1 is an example of the preliminary plots from which such information as is given in Figures 2, 3, and 4 was derived. In Figure 1, brake mean effective pressures are plotted against fuel consumption in pounds per hour. Each curve shows results of a series of runs all made at the same altitude but with different fuel-air ratios. The figure shows curves obtained with the engine operating at atmospheric pressures corresponding to those prevailing at sea level and at altitudes of 5,000, 10,000, 15,000, 20,000, 25,000, and 30,000 feet. Similar groups of curves were obtained with other compression ratios with the engine operating at the same speed, throttle opening, and air temperature. From the curves it was possible to select values of brake mean effective pressure developed with each compression ratio at a given fuel-air ratio. Such values are plotted in Figures 2 and 3. Figure 4 shows similar results obtained at an air temperature of −10° C. In order to pick from such curves as are shown in Figure 1 values corresponding to a given fuel-air ratio, the air flow for the conditions represented must be known. This information is given in Figure 15, which will be discussed later in this report. Many of the figures contain a curve of pounds of fuel per hour versus air density. This curve is merely a record of the fuel consumptions at which the mean effective pressures shown above it were developed. It does not indicate differences in fuel flow produced by differences in air density but on the contrary differences due to adjustments of the carburetor made when the changes in air density took place. The purpose of such adjustments was to maintain the desired fuel-air ratios. Figure 3, for example, shows that when the carburetor was adjusted to give a fuel flow of 58.8 pounds per hour at an air density of 0.045 a brake mean effective pressure of 69.8 was obtained with the 5.3 ratio, 73.8 with the 6.3, 76.8 with the 7.3, and 78.5 with the 8.3.

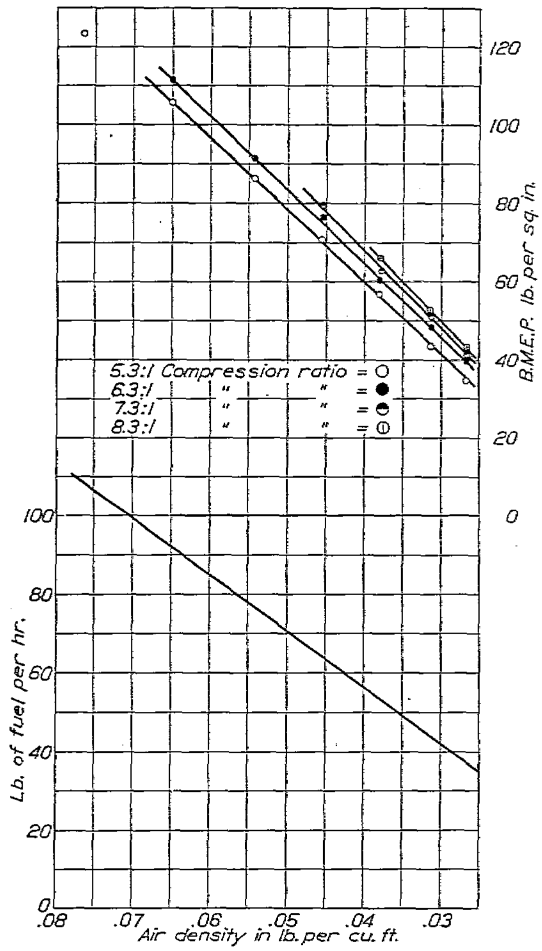


FIG. 2.—Tests 163, 164, 165, and 166. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders; full throttle, 1,600 R. P. M.; air temperature, +10° C.; pound of air per pound of fuel, 13; pound of fuel per pound of air, 0.077

DISCUSSION OF RESULTS

BRAKE HORSEPOWER—POUNDS OF FUEL PER BRAKE HORSEPOWER HOUR

Figures 2, 3, and 4 do not show mean effective pressures developed with the higher compression ratios at the higher air densities (lower altitudes). This is because detonation and preignition prevent satisfactory operation under these conditions with gasoline as a fuel. It was known that the employment of special fuels or throttling of the engine would permit safe sea-level operation even with the 8.3 compression ratio, but it appeared logical first of all to find how much gain resulted from the use of the high compression ratios at high altitudes and then to determine whether or not the gain justified the difficulties incident to the employment of special fuels for operation at low altitudes.

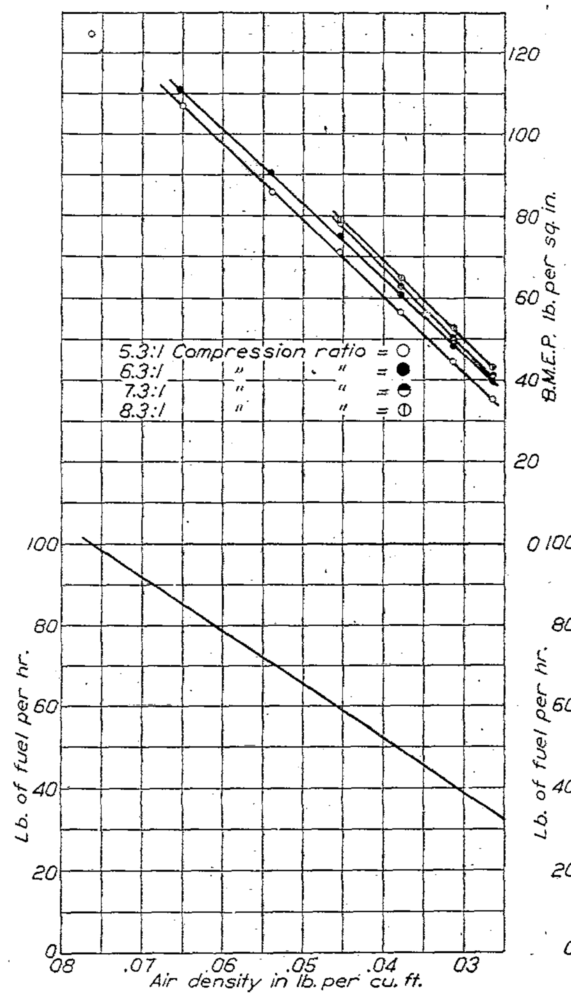
In so far as the performance of this engine over the range of conditions covered by the first four figures is concerned, there was an increase in brake horsepower [2] with each increase of compression ratio. That there was a corresponding decrease in the fuel consumption in pounds per brake horsepower hour is evident from the fact that these figures show the increase in brake horsepower to have been obtained with no increase in the pounds of fuel used per hour. Figures

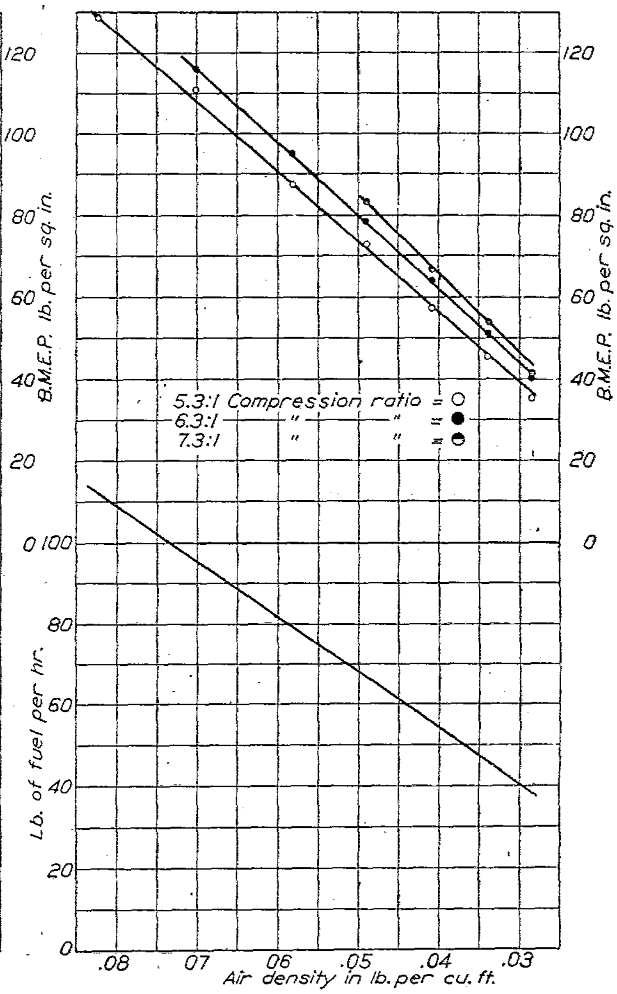
[2] For a given speed and engine, brake horsepower is directly proportional to the brake mean effective pressure.

5 and 6 show increases in brake horsepower obtained with two other engines when their compression ratios were increased from approximately 5.5 to 6.5.

INDICATED HORSEPOWER AND FUEL CONSUMPTION

The original plan of procedure was first of all to obtain and make generally available the information given in the preceding paragraph. This accomplished,[3] the next step was to find the extent to which this information was applicable. Would the benefits derived from an





Tests 163, 164, 165, and 166

FIG. 3.—Air temperature, +10° C.; pound of air per pound of fuel, 14; pound of fuel per pound of air, 0.071

FIG. 4.—Air temperature, −10° C.; pound of air per pound of fuel, 13; pound of fuel per pound of air, 0.077

Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders; full throttle, 1,600 R. P. M.

increase in compression ratio be the same at various engine speeds, at various air temperatures, at various proportions of full power? The basic consideration in an analysis of this sort is the power developed in the engine cylinder, the indicated horsepower. Figures 7 and 8 show an increase in indicated power with each increase in compression ratio. From considerations of air cycle efficiency [4] the increases in power to be expected from increasing the compression ratio from 5.3 to 6.3, 7.3, and 8.3 would be approximately 3 per cent, 9 per cent, and 12 per cent,

[3] Compression Ratio and Thermal Efficiency of Airplane Engines Journal of Society of Automotive Engineers, May, 1921.

[4] Air cycle efficiency = $1-\frac{1}{r^{n-1}}$ where r is the expansion ratio (which equals the compression ratio in conventional engines) and n is the ratio between the specific heat at constant pressure and that at constant volume which for air is approximately 1.4. The derivation of this relation is given in nearly all textbooks on thermodynamics.

respectively. Under the conditions specified in Figure 7 about 96 per cent of the theoretical gain in power was obtained with the 6.3 ratio, 97 per cent with the 7.3 ratio, and 95 per cent with the 8.3 ratio. Under the conditions of Figure 8, 97 per cent of the theoretical gain was obtained with the 6.3 ratio and 94 per cent with the 7.3. Interest attaches to these figures not because they show less than the theoretical gains to have been obtained but because they do show that the ratio of the actual to the theoretical gain is not greatly different for a change in ratio of from 7.3 to 8.3 than for a change from 5.3 to 6.3. In other words the results in no way discourage experiments with even higher compression ratios.

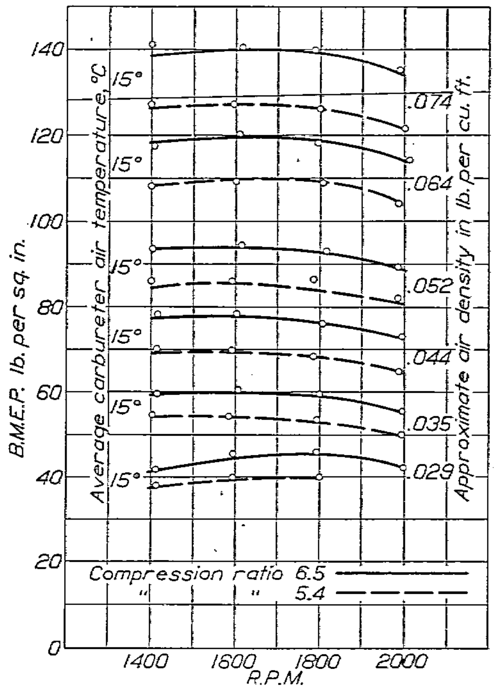


FIG. 5.—Tests 186 and 187. Engine: Bore, 5.51 inches; stroke, 5.91 inches; 8 cylinders

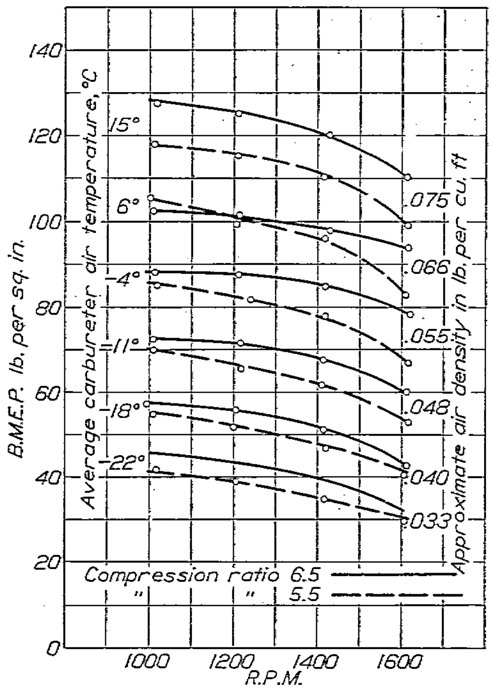


FIG. 6.—Tests 183 and 184. Engine: Bore, 6.625 inches; stroke, 7.5 inches; 6 cylinders

EFFECT OF ENGINE SPEED

Figure 9 is informative as to the influence of engine speed on the gain in power to be anticipated from an increase in compression ratio. The quantities plotted were obtained by dividing indicated mean effective pressures by the appropriate air cycle efficiencies. If the same percentages of air cycle efficiency were obtained with each ratio the curves for the four ratios would coincide. The feature of interest in these curves, however, is not the extent to which the curves coincide but the extent to which a constant relationship between the curves is maintained over this range of speeds.

The curves in the lower group, based on results obtained with an air density of 0.032, are nearly parallel but at the higher densities (lower altitudes) the increase in power with increase in compression ratio appears to be less at low speeds. It seems plausible that this effect may have the same cause as that which makes an engine more prone to detonate or preignite at low speeds. There have been many theories as to the nature of this cause [5] but exact knowledge is lacking. This same condition is evident in Figure 10. The marked difference between the curves for an air density of 0.066 is undoubtedly due to the presence of detonation or preignition with the higher ratio. A blend of benzol and gasoline was used at an air density of 0.075 and as a result the curves at that altitude are almost coincident. Even at the lowest air density, 0.033, the gain in power from the increase in compression ratio is less at 1,000 revolutions per

[5] "The Background of Detonation." Technical Note No. 93 of the National Advisory Committee for Aeronautics, 1922.

minute than at the higher engine speeds. It is quite certain that there was no detonation or preignition at this altitude. Figure 11 shows for another engine, practically no changes in the relation of the 5.4 and 6.5 ratios over a speed range of from 1,400 to 2,000 revolutions per minute. From the evidence here assembled it does not appear that the effect upon engine power of a change in compression ratio varies consistently with changes of engine speed. There is an indication, however, that with some engines increasing the compression ratio increases the power less at low speeds than at high speeds.

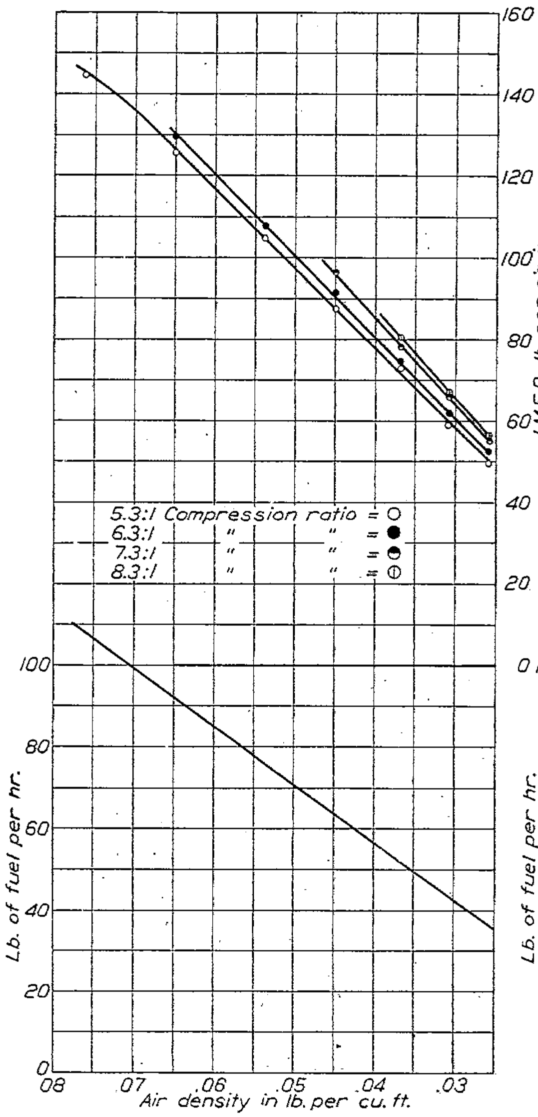


FIG. 7.—Air temperature, +10° C.

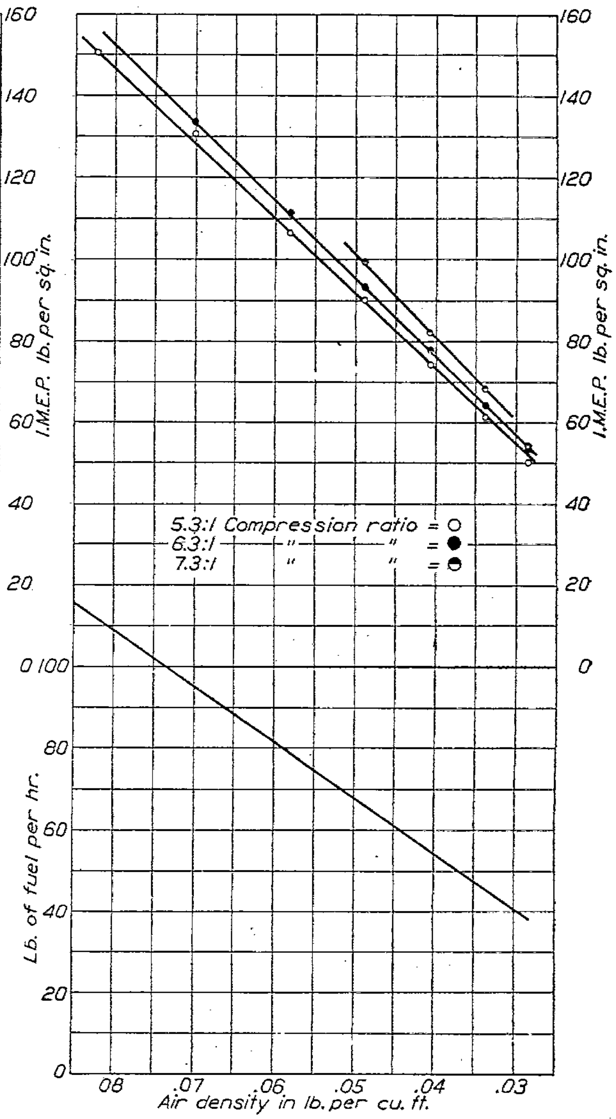


FIG. 8.—Air temperature, −10° C.

Tests 163, 164, 165, and 166. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders; full throttle, 1,600 R. P. M.; pound of air per pound of fuel, 13; pound of fuel per pound of air, 0.077

EFFECT OF ENGINE LOAD

To what extent does the gain in efficiency resulting from an increase in compression ratio depend upon the power developed by the engine? Will it be the same regardless of whether the engine is operating at sea level or at an altitude of 25,000 feet, at full load or at half load? Figure 12 [6] answers these questions fairly satisfactorily. It shows that with a constant fuel-

[6] Similar curves for other compression ratios are given in Report No. 189 of the National Advisory Committee for Aeronautics, entitled "Relation of Air-Fuel Ratio to Engine Performance." 1924.

air ratio the specific fuel consumption (pounds of fuel per indicated horsepower hour) remains practically constant for engine operation over altitudes ranging from sea level to 25,000 feet. It also shows the specific fuel consumption at constant speed to be practically the same for one-half load as for three-fourths load. Obviously, if engine efficiency remains practically constant with change in load then the relation between the efficiencies obtained with different compression ratios will be nearly independent of load. For further illustration, another group of measurements was collected, the specific fuel consumption in this case being measured when the fuel-air ratio had been leaned until the power was 99 per cent of its maximum value. This 99 per cent value has been found to be quite definite and a satisfactory basis for comparing values of specific fuel consumption. The results obtained from these measurements follow:

Compression ratio 5.3
Engine speed, revolutions per minute.... 1,800
Carbureter air temperature .... 40° C.

Approximate altitude	Load	Approximate indicated mean effective pressure	Pound of fuel per indicated horsepower hour at 99 per cent of maximum power
Feet			
5,000	Full	117	0.42
5,000	Part	97	.40
5,000	do	77	.41
25,000	Full	59	.42

Here again it will be noted that the specific fuel consumption remains practically constant regardless of the changes in power brought about by changes in altitude or throttle opening.

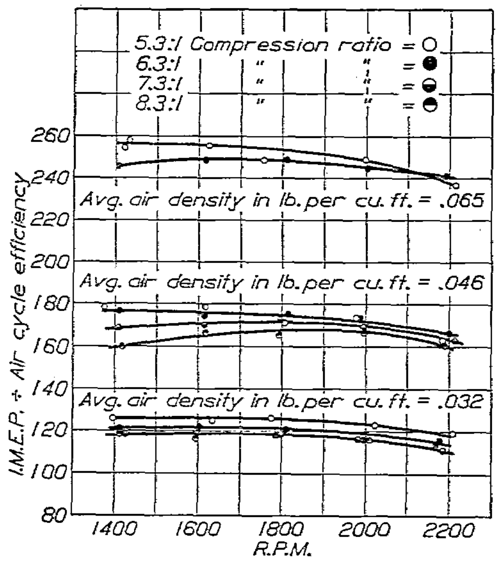


FIG. 9.—Tests 163, 164, 165, and 166. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders; compression ratios, 5.3, 6.3, 7.3, and 8.3. Average carbureter air temperature, +10° C.

Air cycle efficiency $=1-\frac{1}{R^{0.4}}$

Air cycle efficiency $=0.487$ when $R=5.3$
Air cycle efficiency $=0.521$ when $R=6.3$
Air cycle efficiency $=0.548$ when $R=7.3$
Air cycle efficiency $=0.571$ when $R=8.3$

INFLUENCE OF CARBURETOR AIR TEMPERATURES

In order to determine whether or not a change in air temperature would change the relative performance of the engine with various compression ratios, tests were made with a range of air temperatures of from -20 to $+40°$ C. These tests were made at various altitudes, engine speeds and throttle openings. Results of these tests are given in Report No. 190 of the National Advisory Committee for Aeronautics, entitled "Correcting Horsepower to a Standard Temperature." In this report, No. 190, the conclusion is reached that engine power at a constant fuel-air ratio or at a ratio giving maximum power varies nearly inversely as the square root of the absolute temperature regardless of engine speed, engine load, air density, or compression ratio. The report also shows that the thermal efficiency is affected by a difference in temperature only when the change in temperature produces an appreciable change in the amount of fuel that is vaporized in time effectively to enter into combustion. For the aviation engines tested thus far the thermal efficiency has been found to be substantially constant over the range of temperatures noted above provided aviation gasoline is used as fuel. It follows therefore that if the effect of changes in compression ratio upon engine performance be determined at one temperature, the effect upon performance at other temperatures can be predicted with reasonable certainty.

INFLUENCE OF COMBUSTION CHAMBER SHAPE

The method of changing piston head shapes to obtain the various compression ratios has been objected to at times on the grounds that it involved changes in combustion chamber shape which might influence the power development as well as did the change in ratio. It is not unreasonable to expect such an influence, but it is believed that its magnitude is not great. This belief is strongly supported by the single cylinder tests in which combustion chamber shapes ranged between the two extremes shown in Figure 13. In some cases the difference in ratio was effected by a marked change in combustion chamber shape, whereas in other cases the shape was but little altered. This, however, did not appear to influence the closeness with which the change in efficiency approximated the change to be expected from considerations of the air

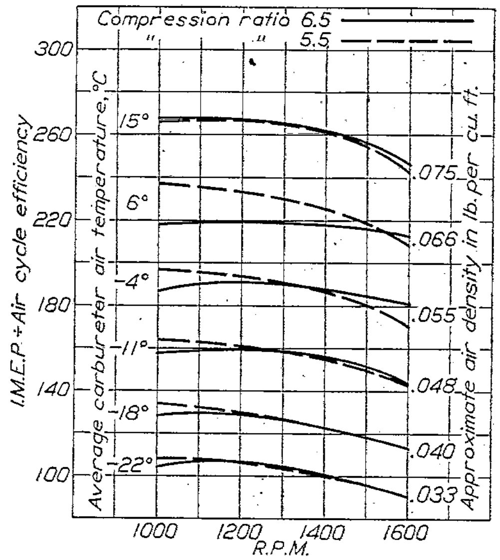


FIG. 10.—Tests 183 and 184. Engine: Bore, 6.625 inches; stroke, 7.5 inches; 6 cylinders. Compression ratios, 6.5, 5.5; air cycle efficiency, 0.527, 0.494

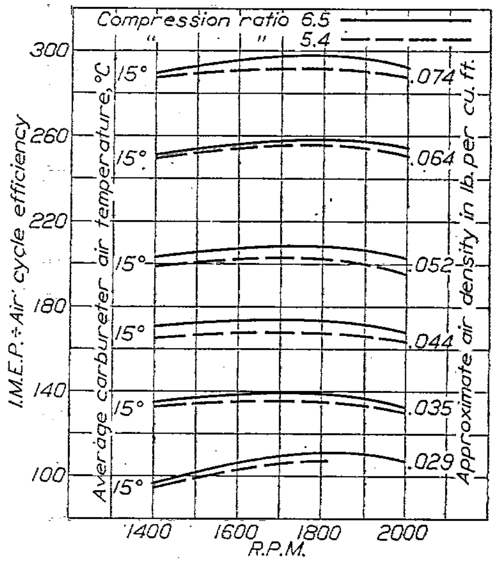


FIG. 11.—Tests 186 and 187. Engine: Bore, 5.51 inches; stroke, 5.91 inches; 8 cylinders. compression ratios, 6.5, 5.4; air cycle efficiency, 0.527, 0.491

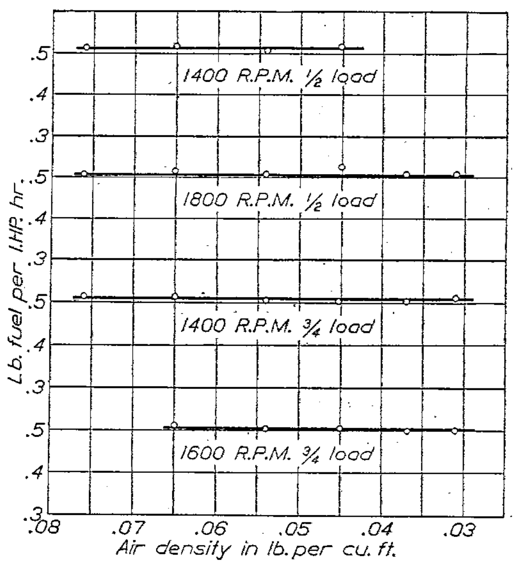


FIG. 12.—Test 168. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders. Compression ratio, 5.3 : 1; air temperature, +10° C.; pound fuel per pound air, 0.08; pound air per pound fuel, 12.50

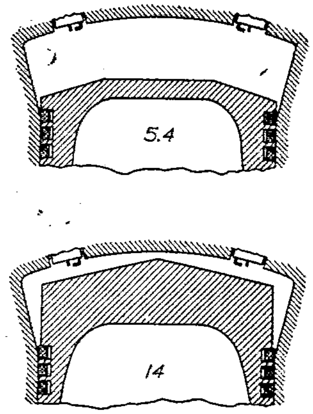


FIG. 13.—Cumbustion chamber shapes

cycle. Hence it does not seem probable that the usually slight modifications in combustion chamber shape made in order to obtain the desired compression ratios should have influenced appreciably the results.

DISCUSSION OF POSSIBLE ADVANTAGES FROM INCREASE OF COMPRESSION RATIO OTHER THAN THOSE WHICH WOULD BE ANTICIPATED FROM A CONSIDERATION OF AIR CYCLE EFFICIENCIES

Among the possible advantages to be discussed are the following:

(*a*) Ability to operate satisfactorily with leaner fuel-air ratios.
(*b*) Increase in volumetric efficiency.
(*c*) Decrease in friction horsepower.
(*d*) Decrease in heat loss to jacket.'

(*a*) ABILITY TO OPERATE SATISFACTORILY WITH LEANER FUEL-AIR RATIOS

If an increase in compression ratio permitted an engine to operate with a leaner fuel-air ratio such an increase would be likely to result in a considerable decrease in the specific fuel consumption of the engine in service use. This arises from the fact that in service a carbureter is likely to be adjusted so that no cylinder receives a mixture too lean to fire under any normal condition of engine operation. Such an adjustment usually gives a mixture that is richer than necessary for most conditions of operation. If it becomes possible to fire leaner mixtures, the carbureter will be adjusted to deliver leaner mixtures and the amount of fuel used in excess of requirements will be less.

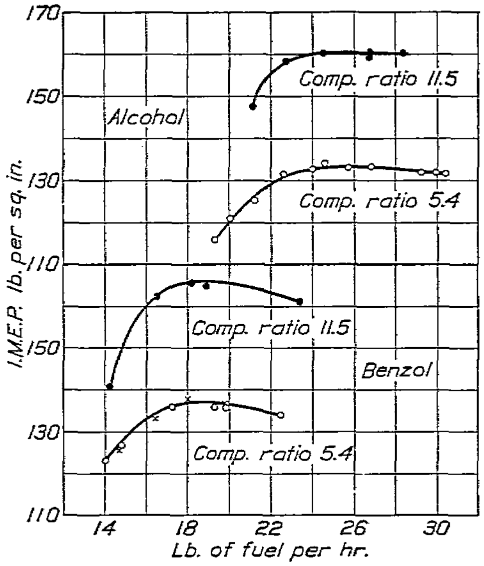


FIG. 14.—Single-cylinder engine tests. Engine: Bore, 5 inches; stroke, 7 inches

There are several reasons for expecting that an increase in compression ratio would permit an engine to be operated satisfactorily with a leaner fuel-air ratio. In the first place because of the smaller clearance volume the ratio of spent gas to new charge will be less. It has been demonstrated experimentally that decreasing the ratio of spent gas to fresh charge does make it possible to fire leaner mixtures. In the second place the temperatures at the end of the compression stroke are somewhat higher for the higher compression ratios. This should tend to make ignition more certain and flame propagation more rapid. Finally one would expect combustion to be completed in a shorter time with the higher compression ratios because the total volume to be traversed by the flame is less. The time required for combustion is of interest since an engine is sometimes unable to operate on a lean mixture solely because the combustion of the mixture is not completed by the time the intake valve opens and as a result the charge in the intake manifold is ignited. None of the tests with multicylinder engines indicated that leaner mixtures could be used with the higher compression ratios. This was far from positive proof that such a condition did not exist, as under many of the conditions of test it was impossible to adjust the carbureter to give a mixture as lean as the leanest upon which the engine could operate. In the single cylinder engine tests to be discussed in more detail later this limitation was not present. Figure 14 shows results obtained with the single-cylinder engine, using compression ratios of 11.5 : 1 and 5.4 : 1 and employing both benzol and alcohol as fuels. Every effort was made to operate the engine with as lean a mixture as possible, although the selection of this mixture is extremely difficult. The curve shows that with both fuels the engine was operated with a slightly leaner mixture with the lower compression ratio. This probably is a consequence of the difficulty in determining exactly the leanest mixture with which the engine can operate, but it rather effectually disposes of the belief that high compression ratios permit engine operation with materially leaner mixtures than low ratios.

(*b*) INCREASE IN VOLUME EFFICIENCY

If an increase in compression ratio increased the volumetric efficiency,[7] namely, the amount of charge entering the engine per cycle, then there would be an increase in power from this source. Published information as to the relation between compression ratio and volumetric efficiency is somewhat contradictory, inasmuch as Ricardo in his single cylinder engine tests found that the volumetric efficiency decreased as the compression ratio was increased, whereas Gibson, another British investigator, found the reverse. With the engine used in Tests 163 to 170, upon the results of which most of this report is based, the air flow was found to be the same within the limits of experimental error for the four compression ratios investigated. Figure 15 gives air flow values obtained at full throttle.

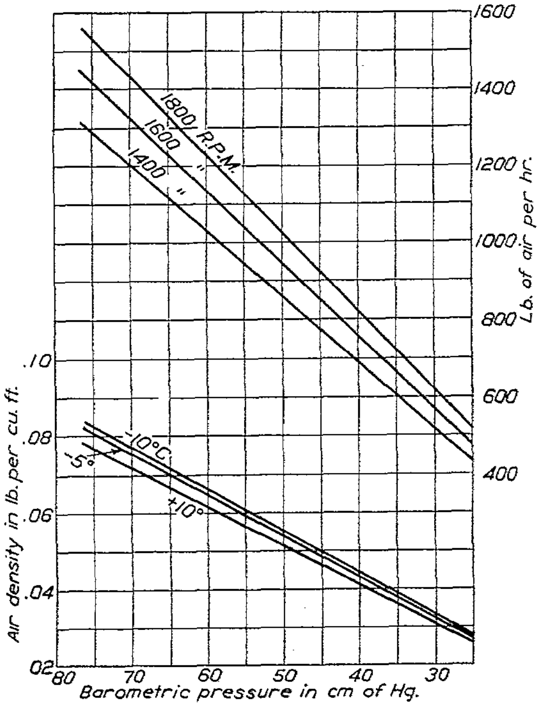


FIG. 15.—Tests 163 to 170. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders. Compression ratios, 5.3, 6.3, 7.3, 8.3. Carbureter air temperature, +10° C. Multiply by 1.036 for −10° C.; multiply by 1.027 for −5° C.

It may be well at this time to point out that it is exceedingly difficult to measure the amount of air entering the carbureter of an engine because of the pulsations in the flow. The quantity of air used by most aircraft engines is too great to make measurement by any positive device such as a displacement meter feasible. It is because of the difficulty in obtaining trustworthy air flow measurements that in much of this report results have been plotted against pounds of fuel per hour rather than against pounds of air per pound of fuel. The same general relations are shown by such plots as it has been found[8] that the change in air flow produced by a change in fuel flow is extremely small. These statements should not be construed as a wholesale condemnation of air-flow measurements but rather as a warning against attributing to such measurements an accuracy which they do not possess. As a matter of fact they probably constitute a satisfactory basis for determining the relative air flow at various air densities, throttle openings, and compression ratios. Such inconsistencies as exist are usually found when making comparisons of air flow at various engine speeds and are a result of changes in pulsation rate.

Despite the difficulties just enumerated it is believed that if any marked change in volumetric efficiency had accompanied the changes in compression ratio, such a change would have been revealed by the air-flow measurements. As has been mentioned, no such condition was found with the engine used in tests 163 to 170.

Figure 16 presents data which support the conclusion drawn from the air flow measurements, although such data hardly constitute a check. The figure is based on compression pressure measurements obtained with a check valve type of indicator. In preparing this figure the chief purpose was to show that the compression pressure at any barometric pressure

[7] As here used volumetric efficiency is the ratio of the volume of air which the engine actually takes in per cycle of two revolutions to the total piston displacement of the engine. This volume is computed for the temperature and pressure existing at the entrance to the carbureter.

[8] In tests at the Bureau of Standards and elsewhere.

could be figured from an equation of the form $P_1 = P_2 R^n$ where P_1 is the compression pressure, P_2 is the barometric pressure, R is the compression ratio and n is an exponent which is nearly constant (for a given type of engine and a given engine speed) for all compression ratios and barometric pressures and which can be determined from the above equation by measuring the compression pressure at one known barometric pressure. Incidentally, however, the fact that with all four ratios the compression pressure could be figured with reasonable accuracy by the use of the same exponent seems consistent with the conclusion drawn from the air flow measurements that the volumetric efficiency was the same for the four compression ratios employed.[9]

With two other engines, one a 6-cylinder of 6.62-inch bore and 7.50-inch stroke and the other an 8-cylinder of 5.51-inch bore and 5.91-inch stroke, the air flow measurements indicated that slightly higher volumetric efficiencies were obtained with a 6.5 : 1 compression ratio than with a 5.5 : 1. These measurements, however, were neither sufficiently complete nor sufficiently consistent to be considered conclusive.

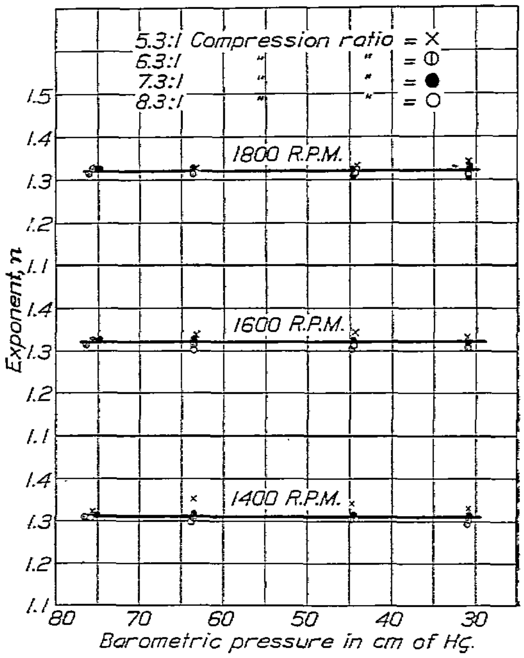


FIG. 16.—Tests 163, 164, 165, and 166. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders. Compression ratios, 5.3, 6.3, 7.3, and 8.3. Curves show value of n in equation $P_1 = P_2 R^n$. P_1=compression pressure, P_2=barometric pressure, R= compression ratio

It seems desirable to discuss briefly some of the reasons for expecting a change in compression ratio to produce a change in volumetric efficiency even though no such effect has proved of major importance in the investigations discussed in this report. The desirability lies in the fact that future changes in engine design may cause influences now insignificant to become of real consequence.

At the end of the suction stroke, the total volume above the piston is the sum of the piston displacement and clearance volume. Part of this space is occupied by the spent gases which originally filled the clearance and part by the new charge.

Let K be the ratio between the volumes occupied by the spent gases at the end and beginning of the suction stroke. The volume occupied by the spent gases at the beginning of the suction stroke is, of course, the clearance volume. The volume occupied by the fresh charge at the end of the suction stroke may be expressed as follows: Volume of fresh charge at end of suction stroke = piston displacement + clearance volume $-$ K (clearance volume) = piston displacement + $(1-K)$ clearance volume.

This equation assumes no heat interchange to have taken place between the spent gases and fresh charge. The assumption simplifies the equation and is justified by the fact that the heat interchange between the gases does not affect the volumetric efficiency as the decrease in volume of the clearance gases due to the heat given up to the entering charge is balanced by the increase in volume of the latter.[10] Referring again to the above equation it is obvious that if K is greater than 1, then $(1-K)$ will be negative and the volume of fresh charge at the end of the suction stroke will be less the greater the clearance volume. In other words the lower the compression ratio (the greater the clearance volume) the lower will be the volumetric efficiency. If K is less than 1 the reverse will be true.

[9] The equation is really an empirical one which by the use of the experimentally determined exponent n permits the compression pressure to be calculated from the atmospheric pressure with approximately the same accuracy as would be possible if the actual pressure at the beginning of compression and the true exponent of compression were known.

[10] This is true only if the fuel is all vaporized before it enters the engine. If this is not the case there is likely to be an increase in volumetric efficiency as the volume occupied by the vapor is less than the decrease in volume of the clearance gases due to the heat abstracted in forming this vapor. The equation, however, is valid for estimating the relative volumetric efficiencies of two compression ratios provided there is no difference in the rate or amount of vaporization that takes place in the engine cylinder during the suction stroke.

For a perfect gas $= \frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2}$ where p is pressure per unit area, v is volume, and T is absolute temperature. This law holds very nearly true for the gases used in the engine, so $\frac{V_2}{V_1} = \frac{p_1}{p_2}\frac{T_2}{T_1} = K$. In this case p_1, v_1, and t_1 represent conditions at the beginning of the suction stroke and p_2, v_2, and t_2 conditions at the end of this stroke. T_2 is usually less then T_1 as there is some heat loss from the spent gases to the surrounding walls during the suction stroke. The ratio of T_2 to T_1 is therefore usually less than 1. In the conventional engine p_2 is ordinarily less than p_1 and the ratio of p_1 to p_2 is greater than 1. A ratio of p_1 to p_2 less than 1 might occur if the intake gases exerted a considerable ramming action and would be very apt to occur when using a mechanically driven supercharger.

As far as the theoretical engine cycle is concerned the exhaust temperature is higher the lower the compression ratio. The higher this tempearture the lower will be the ratio $\frac{T_2}{T_1}$ inasmuch as the loss of heat to the water will increase. Preignition even in its mildest form very materially increases the temperature of the exhaust and as the tendency to preignition increases with the compression ratio it would not be surprising to find the influence of this ratio upon $\frac{T_2}{T_1}$ the reverse of what would be anticipated from the air cycle.

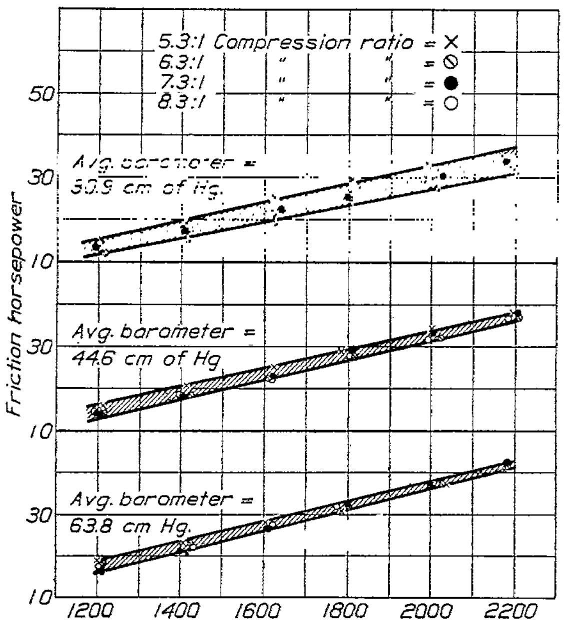


FIG. 17.—Tests 163, 164, 165, and 166. Engine: Bore, 4.72 inches; stroke, 5.12 inches; 8 cylinders. Compression ratios, 5.3, 6.7, 7.3, 8.3. Approximate jacket-water outlet temperature, 40° C.

The influence of compression ratio upon volumetric efficiency under certain circumstances may be rather intimately related to the time of intake valve closing. In nearly all engines the intake valve does not close until the piston has traveled some distance on the compression stroke. If, for two engines differing only in compression ratio, the pressures at the end of the suction stroke are equal, then at the time the intake valve closes the pressure in the engine with the higher compression ratio will be greater than in the other. The greater this pressure the lower at that time will be the rate of flow into the engine cylinder or if this pressure be greater than the manifold pressure the higher will be the rate of flow *from* the cylinder to the manifold. In other words this is one tendency toward a decrease in volumetric efficiency with an increase in compression ratio.

The gist of the foregoing is that experiments to date have shown no major difference in volumetric efficiency to result from changes in compression ratio and that theoretical considerations indicate that such a change may increase or decrease the volumetric efficiency according to the circumstances.

(c) DECREASE IN FRICTION HORSEPOWER

It is *brake* horsepower and *brake* thermal efficiency that is of ultimate concern in any automotive engine. Both increase as the friction horsepower decreases, so that if an increase in compression ratio decreases the friction this gain is as valuable as that due to an increase in air cycle efficiency. Friction horsepower data obtained in these tests are open to the objection that different pistons and rings were used with each compression ratio, offering a possibility for differences in friction to occur as a result of differences in piston or ring clearances. It is to be expected, however, that such differences were rather small inasmuch as every effort was made to make all clearances alike. In Figure 17 are shown friction horsepower measurements for four compression ratios over a speed range of from 1,200 to 2,200 revolutions per minute. The runs showing the greatest differences between ratios—those made at a barometric pressure of 30.9 cm. Hg.—indicate that the highest friction was obtained with the 5.3 ratio and the lowest with the 6.3, the values for ratios of 7.3 and 8.3 being in between. This inclines one to the belief that the differences noted above are due to other causes than the differences in compression ratio. When one appreciates that a change in jacket water temperature of 20° C. often changes the friction horsepower 15 per cent it is easy to understand how changes in friction of the magnitude shown might result from a slight change in oil viscosity. This is the more easily credited when the fact is stated that several months often elapsed between friction measurements with

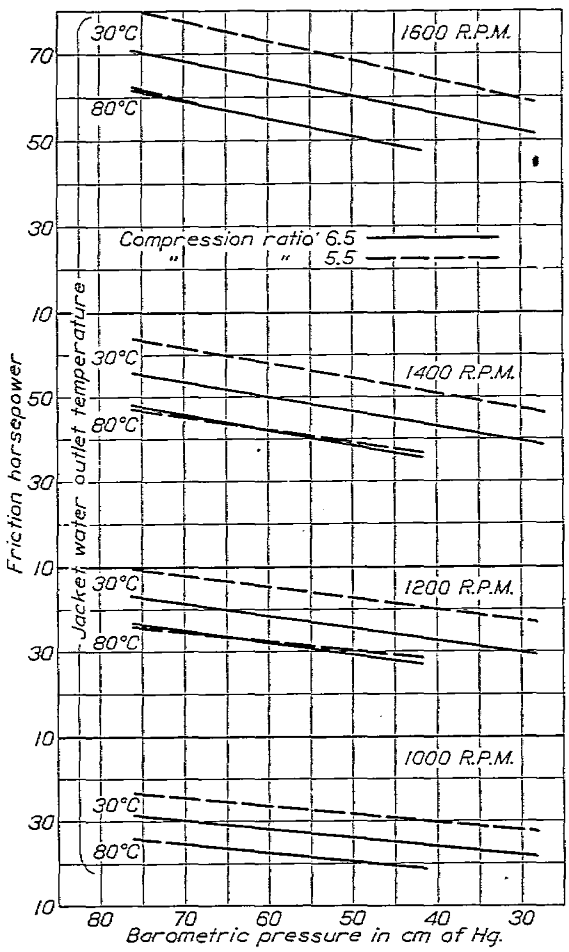


FIG. 18.—Tests 183 and 184. Engine: Bore, 6.625 inches; stroke, 7.5 inches; 6 cylinders

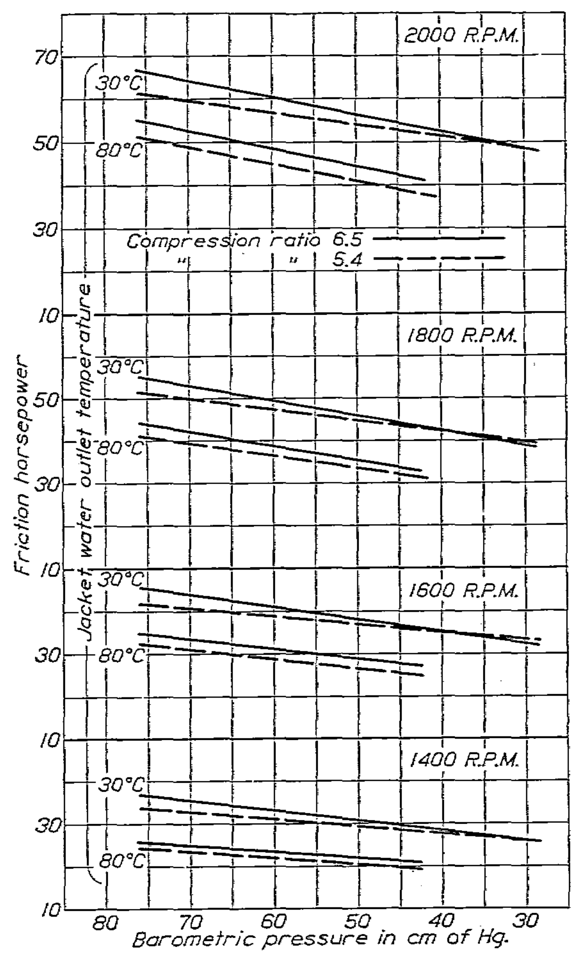


FIG. 19.—Tests 186 and 187. Engine: Bore, 5.51 inches; stroke, 5.91 inches; 8 cylinders

one compression ratio and with another. Figures 18 and 19 show friction data for two other engines. A rather unexpected condition is shown in Figure 18, the friction horsepower being very nearly the same for both compression ratios at a jacket water outlet temperature of 80° C., whereas at a jacket temperature of 30° C. there is a marked difference between the friction horsepower obtained with the two ratios, the lower having the higher friction. In Figure 19 it will be noted that with both jacket water outlet temperatures there was a difference between the friction horsepower obtained with the two compression ratios. Here, however, the highest friction was obtained with the highest compression ratio. In so far as tests with a single cylinder engine having a bore of 5 inches and a stroke of 7 inches are concerned, no changes in friction with change in compression ratio were noted over a range of ratios extending from 5.4 : 1 to 14 : 1.

Figure 20 shows theoretical indicator diagrams for compression ratios of 5 and 8 and also curves for that portion of the side thrust which is due to gas pressure. (The side thrust due to inertia is unaffected by changes in compression ratio.) The side thrust is based on the connecting rod crank ratio of the engine used in these tests. Although this figure shows somewhat higher pressures to be obtained with the higher compression ratio one would not expect the difference in friction to be very great. All in all it appears rather improbable that material differences in friction horsepower should result from differences in compression ratio.

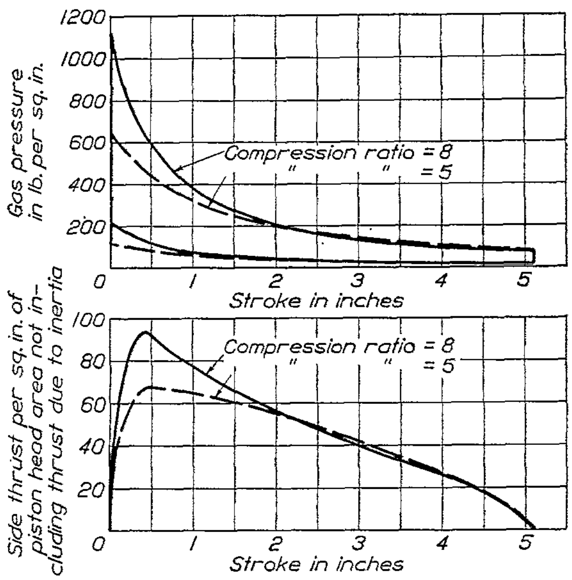


FIG. 20.—Illustrative indicator diagrams for compression ratios of 5 and 8

(*d*) DECREASE IN HEAT LOSS TO THE JACKET

Apart from any possible relation which may exist between the jacket loss and the thermal efficiency of the engine, it is very desirable that the heat loss to the jacket be a minimum. An increase in this loss necessitates more fin surface if the engine be air-cooled or a larger radiator if it be water-cooled. In either case the increase in weight and heat resistance means that more power will be required to propel the airplane.

From theoretical considerations one would expect that the heat loss to the jacket would decrease with increase in compression ratio and that the ratio between the heat loss to the jacket and the heat equivalent of the indicated horsepower would decrease for this reason and also because the engine power increases with increases in compression ratio. Figure 21 shows a plot of gas temperatures over a complete cycle with compression ratios of 4 : 1 and 8 : 1. It will be observed that although with the higher ratio higher temperatures exist at the end of the compression stroke and after combustion, yet because of the high expansion ratio the temperatures rapidly drop during the expansion stroke with the result that the mean temperature over the entire cycle is considerably lower than with the lower ratio. This figure should not be taken as accurately representing the actual temperatures during a cycle, as it is based upon the assumption that no heat is lost to the jacket and upon others but little less bold. If, as the figure shows, the mean temperature during the cycle is considerably lower with the higher compression ratio under the assumption of no loss to the jacket, then it is obvious that the loss which actually does take place must be lower for this ratio unless the surface through which heat transfer takes place is greater. As a matter of fact with the high compression ratio there is less of such surface.

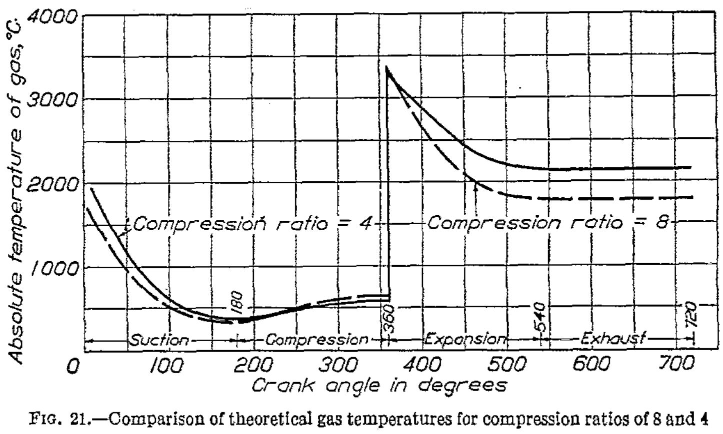


FIG. 21.—Comparison of theoretical gas temperatures for compression ratios of 8 and 4

The data obtained in the compression ratio tests are not consistent enough to justify estimates of the amount of decrease in jacket loss which should result from an increase in compression ratio but they do show conclusively that there is no increase in—

$$\frac{\text{Heat loss to jacket}}{\text{Heat equivalent of indicated horsepower}}$$

And they furnish considerable evidence that there is a decrease. The presence of preignition or detonation greatly increases the heat dissipation to the jacket. Inasmuch as an increase in compression ratio does increase the tendency of an engine to preignite or detonate it may indirectly cause an increase in jacket loss which is the reverse of the normal condition. As stated, this is an abnormal condition, but it is from measurements under such a condition that many have been led to believe that further increases in compression ratio would result in greatly increased jacket losses.

One would expect a decrease in jacket loss to be accompanied by an increase in power and a decrease in specific fuel consumption from an analysis of the data at hand; however, this does not appear to occur to any great extent under normal conditions. For example, there is a marked increase in—

$$\frac{\text{Heat loss to jacket}}{\text{Heat equivalent of indicated horsepower}}$$

with reduction in engine power whether such reduction in power is affected by partly closing the throttle or by reducing the density of the air at the entrance to the carbureter. In spite of the marked increase in this ratio there is no appreciable change in thermal efficiency. The most reasonable explanation of this condition appears to be that most of the heat dissipation to the water jacket takes place during the exhaust stroke. Thermal efficiency is not decreased by the heat lost during this stroke. Why at least half of the heat loss to the jacket should not occur during the expansion stroke is much less easily explained. It may be that the oil film on the cylinder wall serves as a very effective heat insulator during most of the expansion stroke and that it is not until the end of the expansion stroke that this film is burned away and any considerable amount of heat dissipation takes place.

SINGLE CYLINDER TESTS

By the time the tests had progressed this far it had become evident that up to compression ratios as high as 8.3:1 there was a gain in both power and efficiency with increase in compression ratio. The next step appeared to be the extension of this investigation to higher

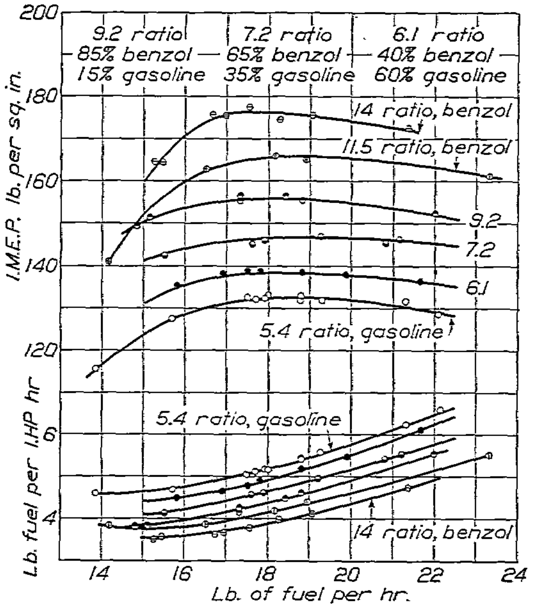


FIG. 22.—Power and fuel consumption. Engine: Bore, 5 inches; stroke, 7 inches; 1 cylinder; 1,500 R. P. M.

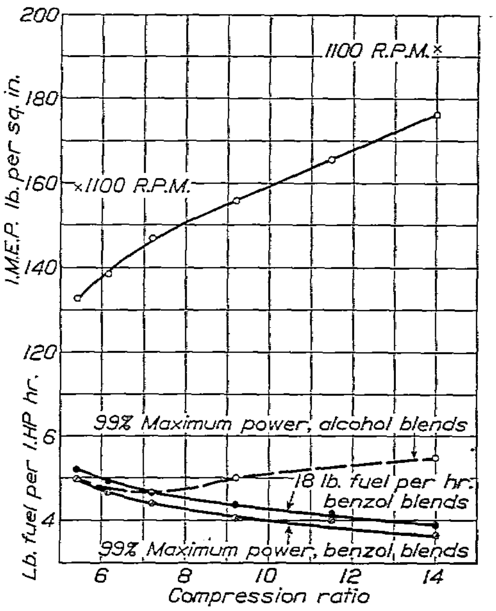


FIG. 23.—Effect of compression ratio on power and fuel consumption. Engine: Bore, 5 inches; stroke, 7 inches; 1 cylinder, 1,500 R. P. M.

compression ratios and for this purpose the single cylinder engine seemed well adapted. This engine was of 5-inch bore and 7-inch stroke, and the changes in compression ratio were obtained by the use of special pistons and by inserting shims between the crankcase and cylinder flange.

Tests were made in the single cylinder engine with compression ratios of 5.4, 6.1, 7.2, 9.2, 11.5, and 14.0. Figure 22 shows results obtained with benzol-gasoline blends, just enough benzol being used in most cases to secure satisfactory operation. No benzol was required with the 5.4 ratio while with the 14 ratio it was necessary to use all benzol. Figure 23 shows the same results but in this case with indicated mean effective pressures and specific fuel consumptions plotted versus compression ratio. These results were obtained at an engine speed of 1, 500 revolutions per minute, but as a matter of interest a few measurements were made at an engine speed of 1,100 revolutions per minute, where the volumetric efficiency of the engine was rather high. At this speed a mean effective pressure of 192 pounds per square inch was obtained with the 14.0 compression ratio.

COMPARISON OF ACTUAL GAINS WITH THOSE WHICH WOULD BE EXPECTED FROM THEORETICAL CONSIDERATIONS

Figure 24 shows the close agreement between the observed increase in efficiency and that which would be expected from a comparison of air cycle efficiencies calculated with an exponent of 1.4. The dotted curve and the points in the figure are derived from actual tests. The tests upon which this curve is based were made under such conditions that there is little likelihood for any appreciable changes in engine condition to have occurred and to have influenced the results. In discussing Figure 7 it was pointed out that although the percentage of air cycle efficiency obtained with ratios higher than 5.3 was less than that obtained with the 5.3 ratio, nevertheless the percentage did not decrease consistently with increase in ratio. From this it would appear probable that a portion of the discrepancy between the actual changes in efficiency and those which would be expected from theoretical considerations were, in this engine, due to changes in engine condition. For the engine whose performance is plotted in Figure 10 it will be noted that the actual gains coincide rather closely with the theoretical except at an engine speed of 1,000 revolutions per minute and at an entrance air pressure corresponding to that at an altitude of 5,000 feet. Probable explanations of these discrepancies have already been discussed. In the case of the engine whose performance is plotted in Figure 11, a higher percentage of air cycle efficiency was obtained with the 6.5 ratio than with the 5.4. The difference, however, is not great. The spark advance of this particular engine was fixed and the same advance was used with both compression ratios.

It may be that this particular spark advance was more suitable for the 6.5 ratio than for the 5.4. In connection with Figure 11 it should be mentioned that a blend of benzol and gasoline was used with the 6.5 ratio for the run at approximately sea-level air pressure and that gasoline alone was used for all the other runs. From a consideration of the results obtained with both the single cylinder and multicylinder engines the conclusion is well justified that over the range of ratios investigated an increase in compression ratio causes an increase in power and thermal efficiency. While the magnitude of these changes is sometimes less than what would be predicted from considerations of air cycle efficiency, nevertheless it is probable that in most cases such considerations will form a fairly reliable basis for estimating the effect of an increase in compression ratio.

OBJECTIONS TO USE OF HIGH-COMPRESSION RATIOS

An increase in compression ratio increases the tendency of an engine to detonate and to preignite. If detonation or preignition becomes serious the gains which otherwise would result from an increase in compression ratio are not realized. Detonation may be defined as a combustion phenomenon whose best recognized manifestation is the ringing sound which usually accompanies a too far advanced spark. Preignition may be defined as ignition from any source prior to the time at which ignition is desired. This ignition may come from self-ignition of the fuel or from some overheated portion of the combustion chamber.

There are at least two distinct methods by which serious detonation and preignition can be prevented. The first method consists in the employment of a special fuel, the second in throttling the engine until satisfactory operation results. Both methods have disadvantages. The subject of fuels for high compression engines will be treated in a future report and hence will be discussed but briefly here. The single-cylinder engine used in this investigation operated satisfactorily with the 14 compression ratio with either benzol or alcohol as fuel. There are several other fuels known to operate satisfactori. y in high-compression engines and more will quite probably be discovered. Unfortunately many of these fuels are inferior to gasoline in other respects. Alcohol, for example, is considerably lower in calorific value than gasoline. Although an engine operating on alcohol develops about the same power as when operating on gasoline, the fuel consumption in pounds per brake horsepower hour is greater. The dotted curve in Figure 23 illustrates the effect of the low calorific value of alcohol. This curve is based upon a series of runs in which blends of alcohol and gasoline were used as fuel, the alcohol content in each case being just sufficient to give satisfactory engine operation. Under these conditions minimum specific fuel consumption was obtained with a compression ratio of about 7 because of the fact that the gain in thermal efficiency resulting from a further increase in compression ratio was offset by the lower calorific value of the blend which it was necessary to employ. It should be observed that this curve is in no sense a contradiction of previous statements that the fuel consumption in pounds per indicated horsepower hour decreases with increase in compression ratio. Had alcohol alone been used at all compression ratios there would have been a decrease in specific fuel consumption with each increase in compression ratio but at all ratios the specific fuel consumption would have been greater than when using gasoline as fuel.

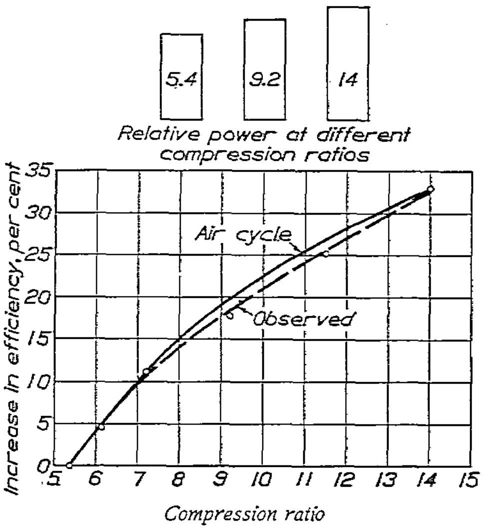


FIG. 24.—Increase of efficiency vs. compression ratio

Throttling an engine for the purpose of employing a high compression ratio also has serious disadvantages as will be more apparent after a consideration of Figure 25. This figure gives a rough basis of comparison of engine characteristics giving equal immunity from detonation troubles with a given fuel. It shows relations between compression ratio, volumetric efficiency, indicated mean effective pressure and compression pressure. Compression pressure curves are based on the equation shown in the figure, P_1 being assumed to be 14.7 pounds per square inch at 100 per cent volumetric efficiency and for lower efficiencies to be lower in direct proportion to the efficiency. The lower group of curves is based upon a measurement of the indicated mean effective pressure and volumetric efficiency of an engine of 5.4 compression ratio. For this measurement the curves in the lower part of the figure were derived by making these two justifiable assumptions: (*a*) That at any compression ratio the indicated mean effective pressure is directly proportional to the volumetric efficiency, and (*b*) that at any volumetric efficiency the indicated mean effective pressure is proportional to the air cycle efficiency as determined by the compression ratio.

An example will illustrate how these curves may be used. Assume a fuel and an engine design such that with an absolute compression pressure of 160 pounds per square inch there is an adequate margin of safety against detonation. The upper curves show a pressure of 160 pounds per square inch to be obtained with a volumetric efficiency of 71.5 per cent at a compression ratio 7 and with a volumetric efficiency of 89 per cent at a compression ratio of 6. The lower curves show an indicated mean effective pressure of 125 pounds per square inch to be developed with a volumetric efficiency of 71.5 per cent at a compression ratio of 7 while an indicated mean effective pressure of 147 pounds per square inch is obtained with 89 per cent volumetric efficiency at a ratio of 6. For the 7 ratio the air cycle efficiency is 54.1 per cent and for the 6 ratio 51.1 per cent. Hence the lower ratio should give 17.5 per cent more indicated power with an efficiency on the basis of indicated power 5.8 per cent less. This figure (fig. 25) presents only the major conditions governing detonation. Size and shape of combustion chamber, location and number of spark plugs, cylinder and piston construction, all these have an influence which usually must be determined by experiment. Moreover, the compression pressure which will provide a constant margin of safety from detonation does increase slightly with increase in compression ratio.[11]

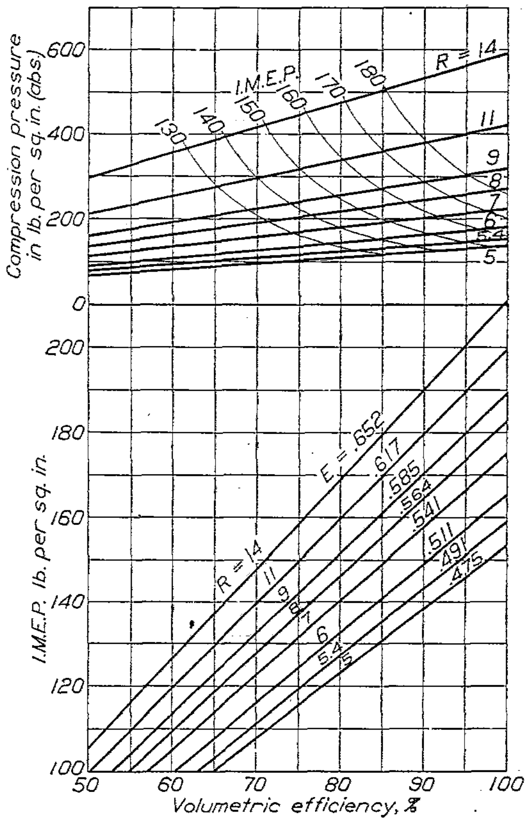


FIG. 25.—Compression ratio, $R=\frac{\text{Displacement}+\text{clearance}}{\text{Clearance}}$ Compression pressure, $P_2=P_1R^{1.4}$. Air cycle efficiency, $E=1-[1/R]^{0.4}$

The disadvantage of throttling an engine to enable a higher compression ratio to be employed does not appear so long as the discussion is confined to indicated horsepower and indicated thermal efficiency. From the preceding discussion, however, it is evident that the throttled high-compression engine must have a greater piston displacement than the unthrottled low-compression ratio engine. Its friction probably is correspondingly higher as piston dimensions, and hence piston friction, usually are proportioned to piston displacement. This increase in friction means a decrease in mechanical efficiency and consequently a decrease in brake horsepower and brake thermal efficiency. Because of the greater piston displacement of the throttled engine its weight will probably be somewhat higher and consequently somewhat higher engine power must be provided to transport this weight. The engine weight, however, is only a part of the total and in the usual airplane it is probable that a 2 per cent increase in engine power would permit a 10 per cent increase in engine weight. It is evident that a considerable amount of calculation or experiment is necessary to determine whether or not an increase in brake thermal efficiency will result from an increase in compression ratio if, in order to employ this ratio, the engine must be throttled. Calculations indicate an over-all gain in brake thermal efficiency of about 2 per cent as the probable result of using a throttled engine of 7 compression ratio instead of an unthrottled engine of 6 ratio. The throttled engine would have the advantage that at altitudes above sea level the throttle opening could be increased and higher power obtained than with the low-compression ratio engine.

It is frequently suggested that by not closing the intake valves until very late on the compression stroke it is possible to have a low-compression ratio and a high-expansion ratio. The objection to this procedure is that the very late closing of the intake valves decreases the volumetric efficiency and hence the power of the engine. Hence the engine is open to most of the objections of the throttled engine and lacks some of the advantages of the latter.

Another disadvantage, which doubtless has suggested itself, is the increase in structural strength or bearing surfaces necessitated by the higher pressures. The maximum pressures encountered in engine operation are those due to detonation and if by the use of special fuels

[11] See "The Background of Detonation," Technical Note No. 93, of the National Advisory Committee for Aeronautics, 1922.

or by other expedients the severity of detonation is not allowed to increase with increase in compression ratio then the maximum pressures should not increase. Mean effective pressures, however, change as the compression ratio changes. The magnitude of the change, however, is rather less than is often supposed. The mean side thrust shown in Figure 20 is 11 per cent higher with the 8 compression ratio than with the 4 and the maximum side thrust 40 per cent higher with the 8 ratio. Figure 20 is based on normal pressures and does not take into consideration detonation pressures which have been discussed above.

It is probably more difficult to start a high-compression engine than one of a lower ratio. This is particularly true of large aviation engines. The starter usually cranks such an engine at rather low speeds and as a result of the low speeds and rather large clearance between pistons and cylinders the leakage past the piston is rather large. Under such conditions the pressure at the end of the compression stroke may be but little higher than atmospheric. At atmospheric pressure the weight of charge in the engine is proportional to the clearance and is lower for the high-compression ratio than for the low. Other conditions being equal the greater the weight of charge fired the more likely the engine is to start. The force of the explosion in the cylinder first fired must be sufficient to overcome the resistance offered by the compression pressure in the cylinder next firing or the engine will not operate. This compression pressure increases with the compression ratio and hence the difficulty in starting increases likewise.

CONCLUSIONS

From the investigation described herein, it is concluded that over the range of ratios investigated an increase in compression ratio produces an increase in power and thermal efficiency provided such a fuel is used that there is no preignition or detonation. The gain in indicated horsepower and indicated thermal efficiency is very nearly that which would be anticipated from a consideration of air-cycle efficiencies. The gain in brake horsepower and brake thermal efficiency should be as great or greater, as there is no consistent evidence that friction horsepower increases with increase in compression ratio. What compression ratio is best for a given engine depends very largely upon what fuel must be employed to avoid preignition or detonation. As has been explained, the disadvantages incidental to the use of some fuels may offset the gain due to an increase in compression ratio.

Further efforts in this field should be directed toward the development of satisfactory fuels for use with high-compression ratios and also toward improvements in combustion-chamber design which will permit higher ratios to be used with a given fuel than is now possible.

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